Robust Semi-Active Ride Control under Stochastic Excitation

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Abstract

Ride control of military vehicles is challenging due to varied terrain and mission requirements such as operating weight. Achieving top speeds on rough terrain is typically considered a key performance parameter, which is always constrained by ride discomfort. Many military vehicles using passive suspensions suffer with compromised performance due to single tuning solution. To further stretch the performance domain to achieving higher speeds on rough roads, semi-active suspensions may offer a wide range of damping possibilities under varying conditions. In this paper, various semi-active control strategies are examined, and improvements have been made, particularly, to the acceleration-driven damper (ADD) strategy to make the approach more robust for varying operating conditions. A seven degrees of freedom ride model and a quarter-car model were developed that were excited by a random road process input modeled using an auto-regressive time series model. The proposed strategy shows promise as a cost-effective solution to improve the ride of a military vehicle over multiple stochastic terrains considering variation in operating weight.

Introduction

There have been many approaches on how to control a vehicle's suspension. The classical method, which is the passive suspension, lacks the ability to adapt to the vehicle's environment. Active suspensions were proposed to provide the ability to tune, in real time, the ride comfort, road holding or any other metric of the suspension behavior that was desired to be tuned. Active suspensions are, however, costly and consume non-trivial quantities of power.

Semi-active suspensions have been available for many years now, and a significant body of research exists on how to utilize them [1, 2, 3]. The most common method of making a suspension semi-active is to use variable value dampers, with constant spring stiffness. The controlled input of the system is the damping value.

One of the many ideas is to utilize classic control techniques such as Linear Quadratic Regulator or other dynamics based control techniques [4, 5]. The problem with employing these techniques is that the control structure, even for a simple model, is nonlinear since the control value, the damping coefficient in this case, is multiplying the velocity of the system, which is a state variable. The technique that is commonly used to bypass this is to define the system as having an active

damper force, and then to clip the control signal so that only the values that the damper can supply are used. The problem with this is that once the optimal control strategy is clipped, it may no longer be as optimal and advantageous with respect to other simpler control strategies. In addition, if nonlinear control is employed, then most likely, full state feedback is required, which is simply not a reasonable assumption. Even if it is known, it can be shown to be inferior to the logic based control systems [6]. Information that can reasonably be known is the relative displacements and velocities across the strut, and the acceleration signals at the top and bottom of the strut. Accurate knowledge of the absolute positions and velocities of the wheel and vehicle body may not be a reasonable assumption due to practical reasons.

Logic based control systems are derived from the kinematics of the vehicle suspension and require no knowledge of the vehicle's parameters. An excellent overview of these is found in [7].

The Accelerometer Driver Damper (ADD) control strategy is very simplistic in operation and requires no knowledge of the vehicle parameters [8]. The lack of knowledge about the vehicle parameters causes some actuator chatter effects in the system. The major contribution of this study is that with some knowledge about the vehicle's parameters the ADD control strategy can be significantly improved. We also show that this control strategy is invariant to the road profile.

Vehicle Modeling

Seven Degree of Freedom Model

Semi-active dampers obey the passivity constraint, such that they only remove energy from the system. To show the effectiveness of the proposed control strategy, a seven degree of freedom (7DoF) vehicle model is implemented as shown in Figure 1. The equations of motion for the seven degree of freedom vehicle model are defined as:

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14. ABSTRACT
Ride control of military vehicles is challenging due to varied terrain and mission requirements such as operating weight. Achieving top speeds on rough terrain is typically considered a key performance parameter, which is always constrained by ride discomfort. Many military vehicles using passive suspensions suffer with compromised performance due to single tuning solution. To further stretch the performance domain to achieving higher speeds on rough roads, semi-active suspensions may offer a wide range of damping possibilities under varying conditions. In this paper, various semi-active control strategies are examined, and improvements have been made, particularly, to the acceleration-driven damper (ADD) strategy to make the approach more robust for varying operating conditions. A seven degrees of freedom ride model and a quarter-car model were developed that were excited by a random road process input modeled using an auto-regressive time series model. The proposed strategy shows promise as a cost-effective solution to improve the ride of a military vehicle over multiple stochastic terrains considering variation in operating weight.

15. SUBJECT TERMS

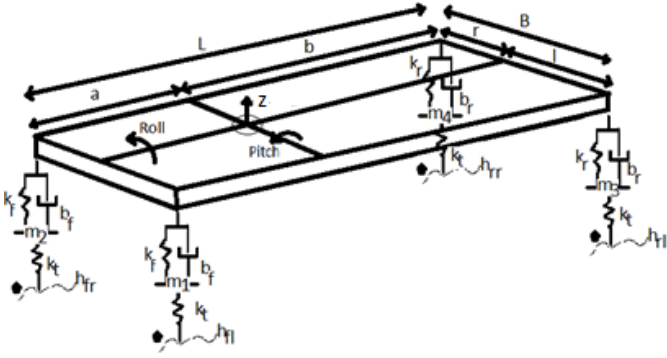| 16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
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Figure 1 7DoF Vehicle Model Diagram

$$F_{fl} = k_{fl}\big(z - a\,\theta + l\phi - z_{fl}\big) + c_{fl}\big(\dot{z} - a\dot{\theta} + l\dot{\phi} - \dot{z}_{fl}\big) \qquad (1)$$

$$F_{fr} = k_{fr}\big(z - a\,\theta - r\phi - z_{fr}\big) + c_{fr}\big(\dot{z} - a\dot{\theta} - r\dot{\phi} - \dot{z}_{fr}\big) \qquad (2)$$

$$F_{rl} = k_{rl}\big(z + b\,\theta + l\phi - z_{rl}\big) + c_{rl}\big(\dot{z} + b\dot{\theta} + l\dot{\phi} - \dot{z}_{rl}\big) \qquad (3)$$

$$F_{rr} = k_{rr}\big(z + b\,\theta - r\phi - z_{rr}\big) + c_{rr}\big(\dot{z} + b\dot{\theta} - r\dot{\phi} - \dot{z}_{rr}\big) \qquad (4)$$

where F_i, k_i, c_i, $i = fl, fr, rl, rr$, are the front right, front left, rear left and rear right suspension force, stiffness and damping value. In relation to the vehicle body positive suspension forces pull the body downward. The geometric properties of the vehicle, a, b, r, l are the distances from the front axle to the center of gravity (CG) of the vehicle, the rear axle to the CG, the right wheels of the vehicle to the CG and the left wheels of the vehicle to the CG. The body coordinates, z, θ, ϕ, are the vertical, pitch and roll displacements of the body. The dot notation represents the derivative with respect to time.

$$\ddot{z}_{fl} = \frac{-k_{u,fl} * \big(z_{fl} - z_{g,fl}\big)H\big(z_{g,fl} - z_{fl}\big) + F_{fl}}{m_{fl}} - g \qquad (5)$$

$$\ddot{z}_{fr} = \frac{-k_{u,fr} * \big(z_{fr} - z_{g,fr}\big)H\big(z_{g,fr} - z_{fr}\big) + F_{fr}}{m_{fr}} - g \qquad (6)$$

$$\ddot{z}_{rl} = \frac{-k_{u,rl} * \big(z_{rl} - z_{g,rl}\big)H\big(z_{g,rl} - z_{rl}\big) + F_{rl}}{m_{rl}} - g \qquad (7)$$

$$\ddot{z}_{rr} = \frac{-k_{u,rr} * \big(z_{rr} - z_{g,rr}\big)H\big(z_{g,rr} - z_{rr}\big) + F_{rr}}{m_{rr}} - g \qquad (8)$$

Equations (5-8) describe the motion of the wheels of the vehicle. $z_i, z_{g,i}$, $i = fl, fr, rl, rr$, are the vertical displacements of the wheels and the ground height at the front left, front right, real left and rear right positions. g is the acceleration due to gravity, defined as $9.81\frac{m}{s^2}$. H is the Heaviside step function that enables wheel-hop. When the wheel's displacement is larger than the grounds displacement this function removes the force that would be generated by the ground, since the wheel is no longer in contact.

$$\ddot{z} = \frac{-\big(F_{fl} + F_{fr} + F_{rl} + F_{rr}\big)}{mass} - g \qquad (9)$$

$$\ddot{\theta} = \frac{a\big(F_{fl} + F_{fr}\big) - b\big(F_{rl} + F_{rr}\big)}{J_{Pitch}} \qquad (10)$$

$$\ddot{\phi} = \frac{-l\big(F_{fl} + F_{rl}\big) + r\big(F_{fr} + F_{rr}\big)}{J_{Roll}} \qquad (11)$$

Equations (9-11) define the vehicle body's equations of motion. $mass, J_{Pitch}, J_{Roll}$ are the mass, pitch inertia and roll inertia of the vehicle body.

In addition to the 7DOF model, a quarter-car model was also developed to validate some of the previous work [7], and demonstrate the effectiveness of the proposed control strategy under similar assumptions.

Stochastic Terrain Model

Time-Series Modeling

In this study we use time-series because it can model both stationary and non-stationary processes. There are three broad classes of time-series models which are of practical importance; the Auto-Regressive (AR) models, the Integrated (I) models, and the Moving Average (MA) models [12]. Combinations of these models result in autoregressive moving average (ARMA) and autoregressive integrated moving average (ARIMA) models. Here, we use autoregressive models.

To model the stochastic road process, Z_g, consider a sample function $u(t) = Z_g$ of a weakly ergodic process which is discretized using a constant time step Δt, such that $u_1, u_2, \dots, u_i$ represent the values at discrete times. An autoregressive model, AR(p), of order p is represented as

$$u_i - \overline{u} = \phi_1(u_{i-1} - \overline{u}) + \phi_2(u_{i-2} - \overline{u}) + \cdots + \phi_p\big(u_{i-p} - \overline{u}\big) + \epsilon_i \qquad (12)$$

where $\overline{u}$ is the mean of the random process $\epsilon_i = N(0, \sigma_\epsilon^2)$, is a Gaussian white noise and ϕ_1, ϕ_2, ϕ_p, are feedback parameters to be estimated depending upon the order p of the AR model. Different order AR models are usually created to determine the best fit. For an AR(2) model, σ_ϵ^2 is determined from

$$\gamma(0) = Var(u_i) = \frac{\sigma_\epsilon^2}{1 - \phi_1\rho_1 - \phi_2\rho_2 - \cdots - \phi_p\rho_p} \qquad (13)$$

Where $\gamma(0)$ is the variance of the random process, and ρ_p is the autocorrelation at the time lag corresponding to p. More details are provided in [12].

After the feedback parameters of the desired model are estimated, the residual series must be tested for uncorrelated, normally distributed white-noise using a series of statistical tests [9,12]. The residual series is the difference between the actual and the estimated processes.

The model type can be identified by visually inspecting the plots of the autocorrelation and the partial sample autocorrelation functions for different lags (multiples of Δt). The autocorrelation provides information on the correlation between random variables $u(t_1)$ and $u(t_1 + h)$ where h denotes the lag. For a stationary random process, the autocorrelation depends only on h and not on t_1. For autoregressive models, the

autocorrelation function dies out quickly with increasing h. The sample autocorrelation function $\hat{\rho}(h)$ is defined as

$$\hat{\rho}(h) = \frac{n^{-1}\sum_{i=1}^{n-h}(u_{i+h}-\hat{\bar{u}})(u_i-\hat{\bar{u}})}{\hat{\sigma}^2} \tag{14}$$

where $\hat{\sigma}$ is the estimated standard deviation of the random process.

The partial autocorrelation represents the autocorrelation between u_i and u_{i+h} without considering the contribution from lags 1 to h-1, inclusive. It is useful in identifying the order of an autoregressive model. The partial autocorrelation of an AR(p) process is zero for lags greater or equal to p+1.

After the order p of the model is identified, the $\phi's$ and $\bar{u}$ are estimated either by using the Yule-Walker equations [9, 12] or equivalently, by minimizing

$$\sum_{i=p+1}^{n}\left((u_i-\bar{u})+\phi_1(u_{i-1}-\bar{u})+\cdots+\phi_p(u_{i-p}-\bar{u})\right)^2 \tag{15}$$

In the current example, a third order Auto-Regressive (AR) model is used to represent the road profile. The equation for the AR(3) model is defined in (16)

$$u_i = \phi_1 u_{i-1} + \phi_2 u_{i-2} + \phi_3 u_{i-3} + \varepsilon_i \tag{16}$$

where u_i is the height of the road profile at each discrete time step, z_r; ϕ_1= 1.2456, ϕ_2 = -0.2976, and ϕ_3 = -0.1954 are feedback parameters estimated from road profile experimental data; and ε_i is Gaussian white noise with mean, µ = 0, and variance, σ² = 0.2634. These coefficients were generated based on number of statistical tests [9,12]. The effect of road variance was studied by varying the variance of the white noise signal, ε_i.

Equation (16) generates a stochastic road profile similar to Figure 2. The x-axis is the longitudinal distance in units of feet and the y-axis is the road height in units of inches. Since the quarter-car model parameters are given in SI units, it is necessary to implement unit conversion in the model. For a vehicle traveling at 20 mph on the stochastic terrain, the sampling rate of the x-axis is 0.0341 to convert the longitudinal distance into time for the simulation. The road input after the unit conversion is shown in Figure 3.

Figure 3 – Final stochastic road profile used in simulation, converted into SI units assuming a vehicle speed of 20 mph.
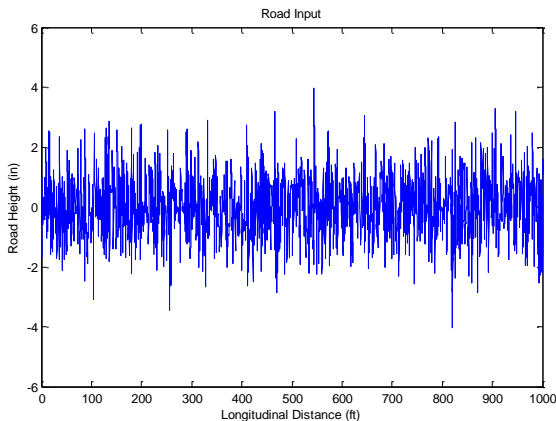
Figure 2 – Initial stochastic road profile generated by AR(3) model

The left and right front wheels of the vehicle model were excited by two independent road samples generated from the stochastic model, from equation (12). The rear wheels were excited following front wheels by the time delay determined by the vehicle speed.
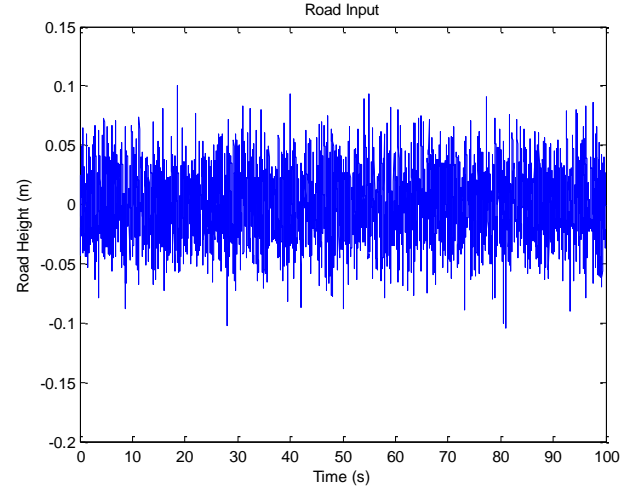

Figure 3 – Final stochastic road profile used in simulation, converted into SI units assuming a vehicle speed of 20 mph.

Ride Comfort Criteria

Absorbed Power

Absorbed power is a measure of the amount of energy a person absorbs from ride vibration over time. It is frequently used in developing military vehicles as a quantitative measure of ride comfort. The average absorbed power is calculated by (17)

$$\overline{AP} = \lim_{T\to\infty}\frac{1}{T}\int_0^T F(t)V(t)dt \tag{17}$$

where $\overline{AP}$ is the average absorbed power, F(t) is the force acting on the person, V(t) is the velocity acting on the person, and T is the averaging time interval [10]. Absorbed power can also be calculated from the sprung mass acceleration using frequency weighting as described in [11].

The actual absorbed power of a person varies with respect the physical characteristics of the individual. Typically, coefficients of a 50th percentile man are used to obtain a single value for comparison. For the 7-DOF model, the absorbed power is computed at all the four seats (two in front and two in rear), and averaged to represent a single ride comfort metric used for the study.

Sprung Mass Acceleration RMS

The RMS of the sprung mass acceleration is used as another measure of ride comfort. It is calculated by equation (18)

$$RMS = \sqrt{\frac{1}{n}\sum_{0}^{n}\ddot{z}_s{}^2} \qquad (18)$$

where $\ddot{z}_s$ is sprung mass acceleration, and n is the number of samples.

Road Holding

Road holding is necessary for safety. If the wheel hop is too high, the vehicle will lose contact with the ground, and lose traction force, making maneuvering difficult or impossible. The wheel hop must be minimized with the goal of maintaining tire force. This is expressed in equation (19)

$$RH = z_u - z_g \qquad (19)$$

where RH is road holding or tire deflection, z_u is unsprung mass displacement, and z_g is road profile height.

Semi-Active Control

Several control techniques are demonstrated here. The ADD control strategy from [7, 8] has been implemented on both the quarter car and the 7DoF model. Improvements have been made to it that significantly increase its performance.

The ADD control strategy is very simplistic in concept. The method can be broken down into four cases in Table 1.

Table 1: Four cases of ADD control strategy

	Body Acceleration	Suspension Force Direction	Desired Damping
1)	Up	Up	C_{min}
2)	Up	Down	C_{max}
3)	Down	Up	C_{max}
4)	Down	Down	C_{min}

These four cases can be written in compact form as:

$$C_{desired} = C_{min} + H(\ddot{z}\dot{z}_{def})(C_{max} - C_{min}) \qquad (20)$$

Where H is the Heaviside step function, $\ddot{z}$ is the acceleration at the top of the strut, and $\dot{z}_{def}$ is the relative velocity of the strut, defined as $\dot{z}_{top} - \dot{z}_{bottom}$.

There are several qualitative arguments for why this control strategy is desirable for comfort control of the vehicle body. Firstly, it reduces power transfer to the body of the vehicle. The power of a point particle can be written as:

$$P = \ddot{z}\dot{z}m \qquad (21)$$

When controlling the acceleration of the system, the velocity of the system is also minimized. SkyHook 2-State control [7], for instance, only minimizes the velocity in the system, and therefore does not minimize the power in the system as effectively, since the acceleration in the system could still be large.

Secondly, if the body is accelerating up and the suspension damping applies an upward force, then to minimize the upward acceleration of the body the smallest damping should be used. Conversely, if the body is accelerating up, but the suspension can supply a negative acceleration then the largest damping term should be used to drive the acceleration toward zero.

The fallacy in this method is that the available damping is never large enough to drive the acceleration to change signs. This is a poor assumption, and it causes the ADD control strategy to produce actuator chatter and jerks to the body's acceleration.

To reduce the actuator chatter and jerks in the system, the control strategy is passed through a moving window filter. The length of the moving window filter is bounded by the damping ratio and the point at which the filtering becomes too large to adequately respond.

Note that the accelerometer used to control the strut's damping force should be mounted at the top of the strut. This configuration makes each strut minimize the acceleration at the top of the strut. Because the dampers obey the passivity constraint, this control technique will globally minimize the acceleration of the body. Conversely, if the acceleration at the CG of the vehicle were used instead of the acceleration at the top of the strut, the vertical acceleration of the CG would be minimized, but the roll and pitch modes may be unaffected, which would not improve the system. This is one of the reasons why the technique is demonstrated on the 7DoF vehicle model.

Moving Window Length Investigation

The ADD control technique is a discontinuous control method that can create significant chatter in the system. To prevent this, a smoothing function should be employed. Here, a moving-average finite-impulse-response (FIR) filter is used. The reason for using this filter is its computationally simplistic nature, though there are other techniques that can be used.

A discrete-time moving-average FIR filter is defined as:

$$Z_k = \sum_{i=0}^{N}\frac{1}{N+1}z_{K-i} \qquad (22)$$

Filter Length Dependence upon Road Surface Properties

The first objective is to show that the control strategy is independent of the road surface. Simulations were performed at various speeds for different road roughnesses and changes in the filter length. The 7DoF vehicle model is used for these tests, and the absorbed power is calculated for each seat in the vehicle. The number is then averaged, and the scale of the z-axis is in watts. The simulation results are shown in Figures 4-6. The sampling rate of the system is 2 kHz.

Each of these figures shows a similar topology. One side of the response is the degradation in performance that is caused by

the actuator chatter. On the other side, the system's performance begins to degrade because too many points are averaged. The choice of the filter length does not correspond to the road input, as demonstrated by the topology of the responses.
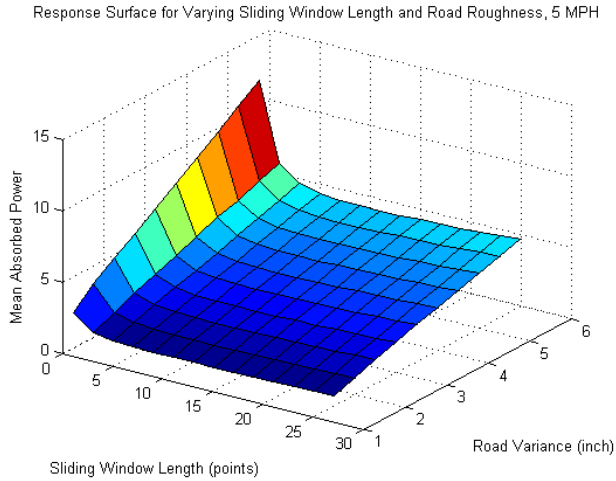

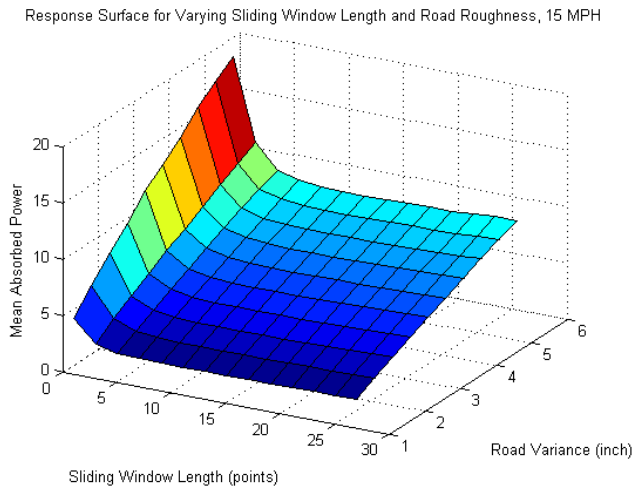Figure 4 – Response Surface for Varying Sliding Window Length and Road Roughness at 5 MPH

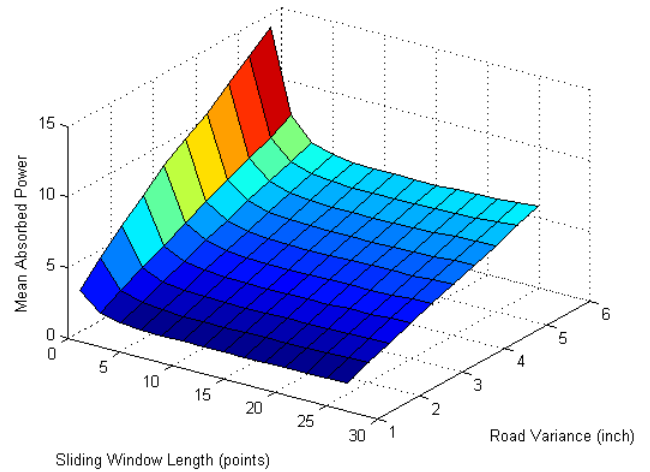Figure 5 – Response Surface for Varying Sliding Window Length and Road Roughness at 15 MPH

Figure 6 – Response Surface for Varying Sliding Window Length and Road Roughness at 30 MPH

The use of the moving-average filter as the smoothing function creates a convex optimization surface. The assumption that the window length can always be increased to compensate for an increase in the damping ratio breaks down for a large number of points, since the system's response bandwidth degrades.

The benefit of this realization is that for systems where the dampers have a slow rate of change, a large number of points will provide a damping value that is optimal for the given terrain the vehicle is traversing. For road surfaces that induce high frequency vibrations, the time average will create a larger average damping value, compared with a lower frequency excitation that will have a lower average value. These choices are in agreement with what is classically known about how to tune a suspension damper for a given excitation frequency.

Filter Length Dependence on Damping Ratio

A set of simulations were performed to assess the interplay between the damping coefficient and the length of the moving average filter.
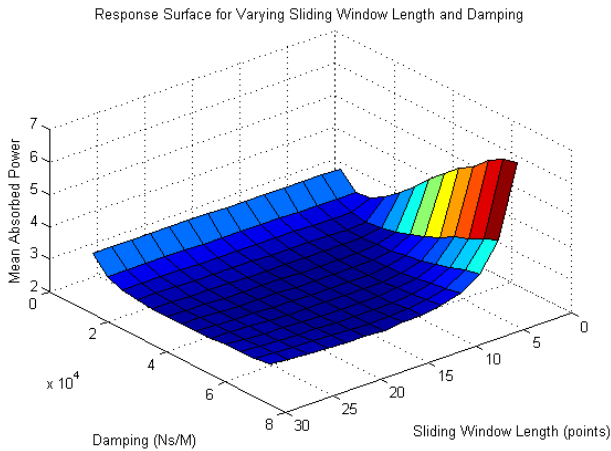
Figure 7 – Absorbed power response for changes in maximum damping and filter length

As shown in Figure 7, changing the damping value or the number of points used in the moving average filter can show significant changes in the response surface. There are three key areas of this response surface that are of particular note. The first is when the damping in the system is too low to be effective at controlling the vehicle's acceleration. The second area is where the damping value is larger, but the number of the moving average filter's points is too small, which causes actuator chatter. The final area is where the number of points for the filter becomes large, and there is non-trivial time averaging. The time averaging of the control signal leads to a reduction in the control laws ability to handle high frequency vibrations. Increasing the length of the moving average will eventually degrade performance. For any given damping, the optimal choice is the 13-point moving-average filter (for a 2kHz sampling rate). The decrease in performance beyond 13 points is very difficult to see in the figures, but it is present in the numbers.

To examine the impact of variations in the vehicle's mass and inertia on the choice of the moving average's filter length, a set of simulations was performed for both the quarter car and the 7DoF vehicle model. For the 7DoF model, the inertia values were set to scale linearly with the mass value. Figure 8 shows the effect of the filter length and the vehicle mass (and inertia) on the absorbed power for the 7DOF model.
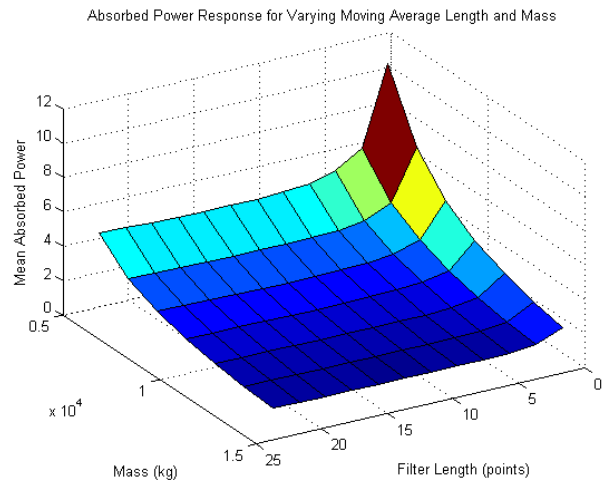

Figure 8 – Absorbed power response for varying filter length and vehicle mass

There are two things to note about this. First, the actuator chatter is present when the filter points are small. Second, the absorbed power decreases as the mass increases. The second observation is expected, because as the mass increases, the vehicle body's acceleration decreases. What was not expected is that there is relatively no change to the optimization surface for the filter length with respect to the vehicle mass.

The qualitative explanation for the latter is because the control law operates partly off of the acceleration of the vehicle body, and as the mass in the vehicle changes it doesn't just change the transfer function, but also how the control law behaves. The proposed control law that is implemented is independent of the vehicle's parameters. Once a damper is selected for a vehicle, using the 13-point moving-average filter on the ADD control system appears to provide the best performance for a very wide range of vehicle parameters.

It should be noted that optimal number of points for the moving average window depends on the sampling rate of the sensors. For the sampling rate of 2000 Hz, the optimal number of points is 13. For lower sampling rates, the number of filter points should decrease. At the given sampling rate, the approximate delay introduced by the moving-average filter should be about 6 ms.

Results

7DoF Control Strategy Results

The 7DoF model is simulated to compare three different control strategies: the original ADD control, the proposed smoothed ADD control, and a passive damping. To examine robustness of the proposed strategy, results are compared considering variations in both road roughness and mass of the vehicle.

Two metrics evaluated are: 1) the absorbed power, which is the average absorbed power across the four seats of the

vehicle, shown in Figures 9 to 11, and 2) the road holding, which is the worst maximum displacement of the four wheels of the vehicle, shown in Figures 12 to 14.
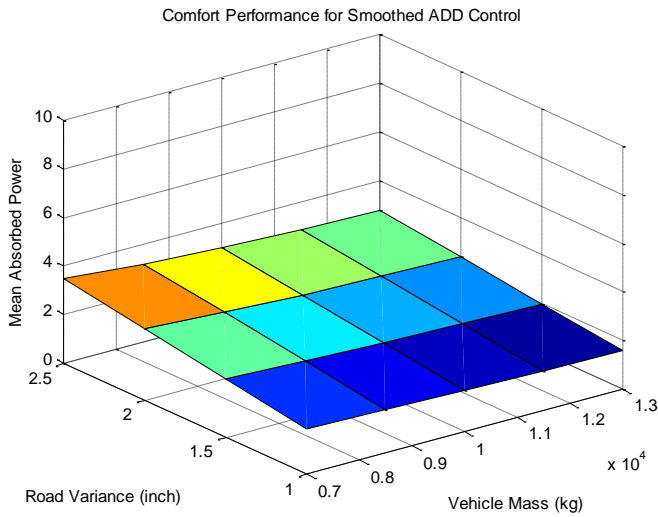
Comfort Performance for Smoothed ADD Control

Figure 9 – Absorbed power for changes in vehicle mass and road roughness using the proposed smoothed ADD control strategy

Comfort Performance for Original ADD Control
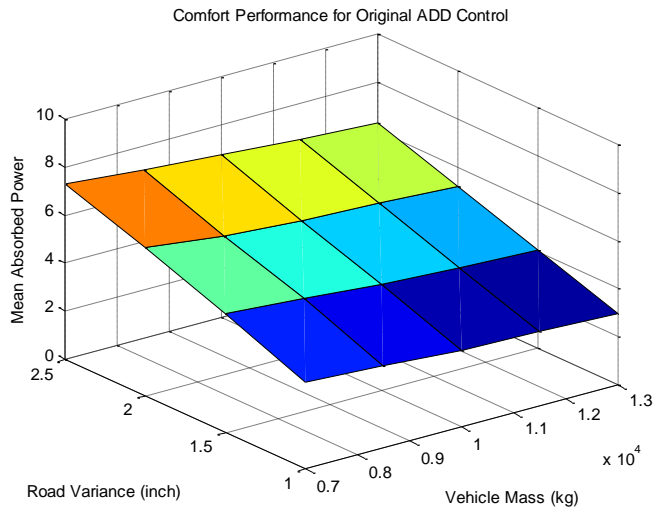
Figure 10 – Absorbed power response for changes in vehicle mass and road roughness for the original ADD control strategy

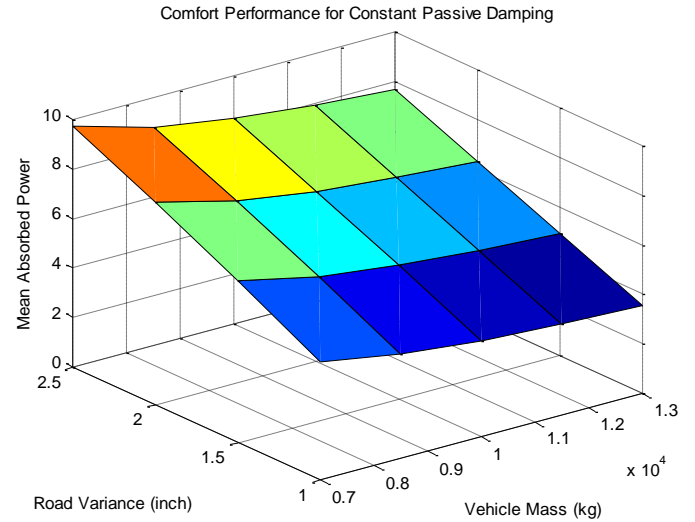Comfort Performance for Constant Passive Damping

Figure 11 – Absorbed power response for changes in vehicle mass and road roughness for a passive damping configuration

As seen in Figures 9 to 11, the proposed control strategy performs significantly better for ride comfort, reducing the absorbed power at the vehicle's seats by more than 50% with respect to that from the original ADD strategy. The original ADD damper strategy improves comfort by about 25% with respect to the passive damper.

The second comparison is for the road holding metric. According to Figures 12 to 14, the road-holding metric is degraded by about 15% due to smoothing, while there is insignificant change in road holding between passive and ADD.
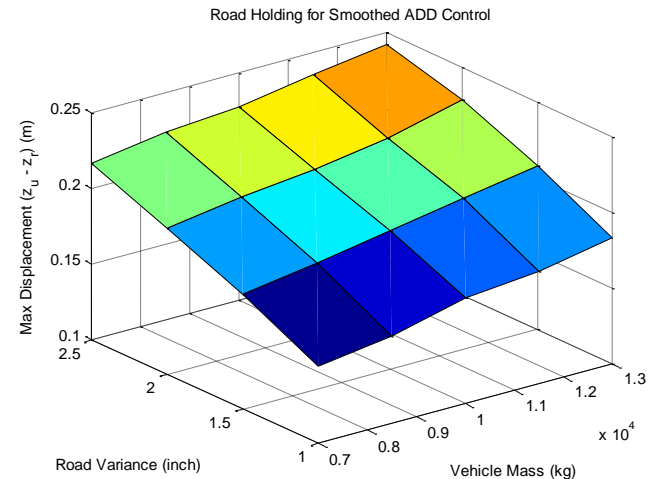
Road Holding for Smoothed ADD Control

Figure 12 – Road holding evaluation for changes in vehicle mass and road roughness using the proposed smoothed ADD control strategy
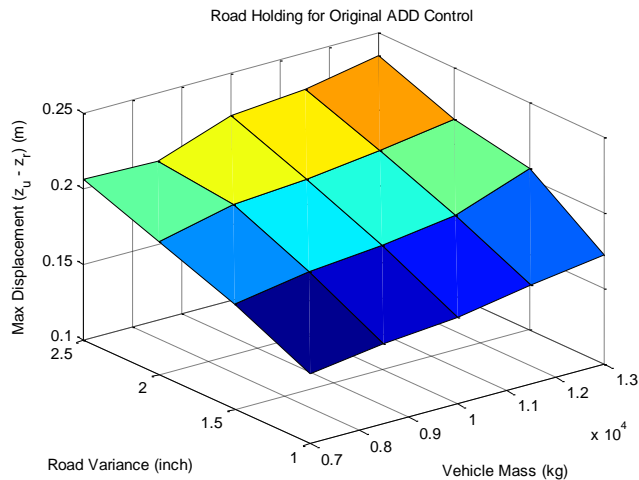
Figure 13 – Road holding evaluation for changes in vehicle mass and road roughness using the original ADD control strategy
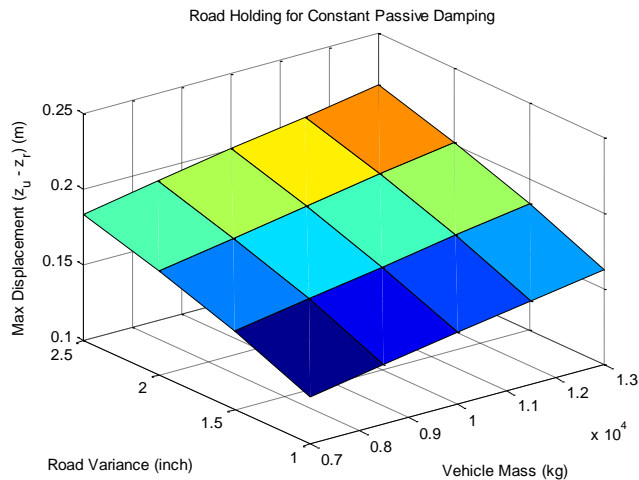

Figure 14 – Road holding evaluation for changes in vehicle mass and road roughness using a passive damper configuration

Quarter-Car Results

In addition to 7DOF, the effectiveness of the proposed smoothed ADD strategy is also demonstrated using a quarter-car model. Though 7DOF would provide a higher fidelity representation of the vehicle performance than a quarter-car model, some of the previous research used a quarter-car model. The stochastic road profile is simulated in the quarter car model for several control strategies including Skyhook (SH) 2-state [12], Skyhook Linear [13], Acceleration Driven Damper (ADD) [8], and Skyhook-Acceleration Driven Damper (SH-ADD) [14] combination. Details of the control strategies can also be found in [7]. Corresponding to the maximum damping ratio in the system, a filter length of 13 points is used for the smoothed ADD control algorithm. The simulation results are shown in Table 2.

Table 2: Results of Quarter Car Simulation

	Average Absorbed Power (W)	Sprung Mass Acceleration RMS (g's)	Road Holding Max (in)
Passive	26.65	0.61	4.45
SH 2-state	6.19	0.39	4.87
SH-ADD	3.43	0.25	4.87
SH Linear	3.05	0.23	5.54
ADD	1.28	0.19	5.11
Smoothed ADD (Proposed)	1.09	0.17	5.18

The quarter car results indicate that the ADD control strategy significantly decreases the average absorbed power while slightly increasing the road holding. The proposed smoothed ADD control strategy further reduces the absorbed power by about 15%, while only increases the road holding metric by about 1.4%. Where the original ADD approach demonstrates significant improvement for the quarter car model, the benefits of smoothing ADD is much higher in the case of the 7DOF model. This may be due to the fact that actuation chatter effect may be more pronounced due to independent control of four corners constrained by the rigid body motion of the sprung mass. Smoothing ADD reduces the impact of constraint by letting damping rate respond slower.

Summary/Conclusions

Presented here is a method for improving the ride comfort of a vehicle. This method has been shown to be invariant with respect to both road surface and the vehicle parameters such as mass and vehicle speed etc. In order to represent road stochastics, a time-series model was developed, and investigated for various road roughnesses. It was found that the original ADD technique works well with a quarter-car model. However, in a higher fidelity 7DOF model, actuation chattering becomes more pronounced, which negatively affects ride comfort. The proposed smoothing ADD technique reduces chattering and provides noticeable improvement over the original ADD control algorithm in terms of absorbed power. The proposed approach is also found to be more robust and less sensitive to design and operational uncertainties.

However, road holding does decrease a small amount in comparison to the original strategy. Cost functions could be put in place to create a tradeoff between road holding and ride comfort for different scenarios, such as cornering or straight-line driving.

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